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1. Your reference	M03B329/ASB	- 9 MAR 2004
2. Patent application numl *(The Patent Office will fill thi*	0405317.9	10MAR04 E879593-1 D02805 P01/7700 0.00-0405317.9 NONE

3. Full name, address and postcode of the or of each applicant *(underline all surnames)*

The BOC Group plc, Chertsey Road, Windlesham, Surrey, GU20 6HJ

Patents ADP number *(if you know it)* 884627002

If the applicant is a corporate body, give the country/state of its incorporation: England

4. Title of the invention: IMPROVEMENTS IN SCREW PUMPS

5. Name of your agent *(if you have one)*: Andrew Steven BOOTH

"Address for service" in the United Kingdom to which all correspondence should be sent *(including the postcode)*

The BOC Group plc, Chertsey Road, Windlesham, Surrey, GU20 6HJ

Patents ADP number *(if you know it)*

6. Priority: Complete this section if you are declaring priority from one or more earlier patent applications, filed in the last 12 months.

Country	Priority application number *(if you know it)*	Date of filing *(day / month / year)*
GB	0326061.9	10/11/2003

7. Divisionals, etc: Complete this section only if this application is a divisional application or resulted from an entitlement dispute (see note f)

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8. Is a Patents Form 7/77 (Statement of inventorship and of right to grant of a patent) required in support of this request? Yes

Answer YES if:
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Continuation sheets of this form	0
Description	9
Claim(s)	3
Abstract	1
Drawing(s)	4 + 4 8

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Priority documents	0
Translations of priority documents	0
Statement of inventorship and right to grant of a patent (Patents Form 7/77)	0
Request for a preliminary examination and search (Patents Form 9/77)	0
Request for a substantive examination (Patents Form 10/77)	0
Any other documents (please specify)	0

11. I/We request the grant of a patent on the basis of this application.

Signature(s) A R.B Date 09/03/2004

12. Name, daytime telephone number and e-mail address, if any, of person to contact in the United Kingdom

Andrew Booth
(01276) 807612

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IMPROVEMENTS IN SCREW PUMPS

This invention relates to screw pumps, more specifically to screw pumps with tapered screw mechanisms and which are typically used in vacuum applications. The invention is directed to improvements in the operational efficiency of the aforementioned pumps.

Screw pumps are widely used in industrial processes to provide a clean and/or low pressure environment for the manufacture of products. Applications include the pharmaceutical and semi-conductor manufacturing industries. A typical screw pump mechanism comprises two parallel spaced shafts each carrying externally threaded rotors, the shafts being mounted in a pump body such that the threads of the rotors intermesh. Close tolerances between the rotor threads at the points of intermeshing and with the internal surface of the pump body (which acts as a stator), causes volumes of gas entering at an inlet to be trapped between the threads of the rotors and the internal surface and thereby urged towards an outlet of the pump as the rotors rotate.

Various adaptations of the basic screw pump mechanism are known, for example, there exist screw pumps with variable pitch screw threads and/or mechanisms wherein the height (or outside diameter) of the screw thread tapers decreasingly in a direction from the pump inlet to the exhaust of the pump. In the latter case, the rotors are mounted in a tapering bore of the stator.

It is desirable when operating a screw pump to achieve a desired pressure ("ultimate pressure") which is typically significantly below atmospheric pressure, that the input power needed to operate the pump is minimised. The size of the pump exhaust volume has a considerable effect on the input power needed to operate a pump at ultimate pressure. The input power can be maintained low at ultimate pressure by inbuilding a high volume ratio between the inlet volume and the exhaust volume of the pump. A disadvantage of this arrangement is that as

the inlet pressure of the pump increases towards atmospheric pressure, there is a significant increase in the input power requirements of the pump.

In the prior art, high pump inlet pressures have been avoided by inclusion of a blow-off valve within the pump which can be activated to release pressure and prevent build up of excessive pressure in the pump. In some situations, the performance of these valves can be adversely affected by build up of process media on or near sealing surfaces, reducing the efficiency with which pressure build up can be relieved.

The present invention aims to provide an alternative and more reliable means for reducing the power input requirements of a screw pump.

In accordance with the present invention, there is provided a screw pump comprising a stator housing first and second externally threaded rotors adapted for counter-rotation within the stator, and means for actively controlling the axial position of the rotors within the stator during use of the pump.

Actively controlling the axial position of the rotors within the stator in real time can significantly reduce the effect of variables such as machining tolerances and running temperatures on pump performance. It not only allows reliable close running of rotor to stator, but also allows this clearance to be increased, decreased or maintained at a constant level as necessary during use of the pump.

For example, in one embodiment the control means comprises means for effecting or resisting axial movement of the rotors in response to an axial load generated in the rotors during operation of the pump. When in operation, internal pressure within a pump produces an axial thrust load in the rotor. This thrust load is proportional to the amount of gas compression work being performed by the pump and hence the input power requirements of the pump. The efficiency of gas compression of a screw pump is, to a large extent, dictated by the clearance between the internal surface of the stator which carries the screw threaded rotors

and the rotors themselves. Where the rotors are tapered, they may be moved both simultaneously and synchronously away from the stator face effectively increasing the radial clearance, reducing the compression and hence the power input requirements.

Each rotor is typically mounted on, or integral with, a respective shaft rotatably mounted within the pump. The pump comprises a bearing assembly for rotatably supporting the shafts relative to the stator, the control means preferably comprising means for moving the bearing assembly relative to the stator.

In one embodiment, a piston engages the bearing assembly, the moving means being arranged to move the piston relative to the stator to control the axial position of the rotors. The piston may conveniently comprise part of a housing for the bearing assembly. The moving means may comprises a motor adapted to rotate a drive shaft which engages the piston so as to axially move the piston relative to the stator with rotation of the drive shaft. The drive shaft may, for example, comprise a lead screw which passes through a conformingly-threaded aperture in the piston. Alternatively, the moving means may comprise one or more electromagnets for moving the piston, or any other convenient mechanism for accurately moving the piston relative to the pumping chamber.

Thus, in this embodiment, the control means preferably comprises a piston actuatable by a control device so as to control the axial position of the rotor. The control device may be responsive to an input power and resulting axial load on a rotor to cause the piston to actuate so as to control the axial position of the rotor. Alternatively, or additionally, the control means may be used to actively move the rotors closer to the stator surface during use of the pump to scrape off process media built up on the stator surface. Furthermore, the control means may also be used to maximise the rotor to stator clearance when the pump is switched off following pumping of sticky or dusty atmospheres, so as to prevent problems occurring upon restart. The rotor to stator clearance may also be controlled to optimise pump performance for different pumped gas species. For example, the

clearance can be increased when pumping hydrogen or an inert gas such as argon so as to achieve optimum performance without pump seizure. The control device may be configured to receive a signal indicative of the temperature of and/or within the stator, and to control the axial position of the rotors in dependence of this signal so as to avoid clashing between the rotors and the stator. One or more other operational parameters, such as ultimate vacuum calibration at start up, back pressure, exhaust temperature, power consumption and inlet pressure, can be used to control the axial position of the rotors.

In another embodiment, the bearing assembly is free to move in an axial direction within a housing, the control means comprising a spring mechanism arranged with respect to a rotor such that when the rotor is subjected to an axial load, the spring mechanism compresses or extends causing an axial reactive load, whereby to maintain a constant axial position of the rotor over time.

When an axial load generated in a rotor tends to cause axial displacement of the rotor and bearing assembly, the spring may be compressed or extended (depending on its position). Assuming the load does not exceed the elastic limit of the spring, the spring will react to vary the axial position of the rotor. By selecting a spring with a suitable spring constant, the arrangement can be used to vary the rotor/stator clearance giving a relatively constant level of gas compression work over a wide range of inlet pressures, thereby moderating the power input requirements of the pump.

The control means are preferably arranged so as to ensure both rotors are maintained in the same axial position. However, the control means may be configured so as to permit relative axial movement between the rotors. Typically, such relative movement will be within the limits of rotor contact and might be used with the rotors in operation to scrape off process media build up on the flanks of the screw threads of the rotors. The latter can be achieved using independent means for effecting axial movement of each rotor, for example respective piston

arrangements as previously mentioned. An associated control device may be configured to actuate the pistons independently of one another.

In the preferred embodiments, at least part of the screw threads of the rotors have an outside diameter which tapers decreasingly in a direction from the pump inlet to the exhaust of the pump. In one embodiment, each screw thread has a diameter which gradually decreases from the pump inlet to the exhaust. In another embodiment, only part of the screw thread of each rotor has an outside diameter which tapers towards the exhaust of the pump, the remainder of the screw thread having a substantially constant diameter. There are a number of advantages particularly associated with this latter embodiment. Firstly, vacuum pump exhaust gas temperatures vary with running conditions, and have an effect on the rotor to stator clearance at the exhaust (low vacuum) end of the pump. Control of the rotor to stator clearance in the exhaust stages allows the optimisation of performance and power consumption. The inlet (high vacuum) temperature does not vary as considerably as the exhaust and hence rotor to stator control in the inlet stages is of lesser importance. Secondly, during roughing (pumping large volumes of gas at or near atmospheric pressure) performance can be optimised by bypassing the low vacuum stages of the pump. The rotor to stator clearance in the exhaust stages can be increased to act as a pressure relief valve, with the rotor to stator clearance at the inlet stages remaining constant so as to maximise pumping efficiency.

By way of example, some embodiments of the invention will now be further described with reference to the following Figures in which:

Figure 1 to 4 show a first embodiment of a screw pump in four different views;

Figures 5 to 7 show in more detail, the means for effecting axial movement of the rotors in the embodiment of Figure 1; and

Figure 8 shows a section through a second embodiment of a screw pump.

Figures 2 and 3 show respectively side and top views of a first embodiment of a screw pump. Figure 1 shows a section through the plane B-B marked in Figure 2 and Figure 4 shows a section through the plane A-A marked in Figure 3.

The pump 10 includes a pump body 12 having a first part 14 and a second part 16 defining a pumping chamber 18. A fluid inlet 20 to the chamber 18 is formed in the first part 14 of the pump body 12, and a fluid outlet 22 from the chamber 18 is formed in the second part 16 of the pump body 12.

The pump 10 further includes a first shaft 24 and, spaced therefrom and parallel thereto, a second shaft 26. Bearings 28 are provided for supporting the shafts 24, 26. The shafts 24, 26 are adapted for rotation about the longitudinal axes thereof in a contra-rotational direction. One of the shafts 24 is connected to a drive motor 30 via a drive mechanism 32, the shafts being coupled together by means of timing gears so that in use the shafts 24, 26 rotate at the same speed but in opposite directions.

A first rotor 34 is mounted on the first shaft 24 for rotary movement within the chamber 18, and a second rotor 36 is similarly mounted on the second shaft 26. Each of the two rotors 34, 36 has a tapered shape and has a helical vane or thread 38, 40 respectively formed on the outer surface thereof, the threads intermeshing as illustrated.

In this embodiment, the screw threads 38, 40 of the rotors 34, 36 have an outside diameter which tapers decreasingly in a direction from the inlet 20 to the outlet 22 of the pump 10, and the inner surface of the pumping chamber 18, which acts as a stator during use of the pump, conformingly tapers towards the pump outlet 22. The shape of the rotors 34, 36 and in particular the shapes of the threads 38, 40 relative to each other and to the inner surface of the pumping chamber 18, are calculated to ensure close tolerances with the inner surface of the pumping chamber 18.

In order to control the axial position of the rotors 34, 36 within the chamber 18, the pump 10 includes bearing assemblies 42 each slidably mounted within a respective cylindrical housing 44, as shown in more detail in Figures 5 to 7, located at the exhaust end of the pump 10. Each cylindrical housing 44 is fastened to a respective shaft 24, 26, the cylindrical housings 44 being connected to each other by means of a connecting arm 46.

Each bearing assembly 42 comprises a pair of angular contact bearings 48 arranged in a back to back configuration to maintain the lateral position of the shaft 24 passing through the bearing assembly 42 with respect to the pump body 12 whilst allowing axial movement of the shaft and rotation of the shaft about its longitudinal axis. A small clearance is provided between the outer surface of the bearings 48 and the inner surface of the cylindrical housing 44. The cylindrical housing 44 also has a small radial clearance with the pump body 12 which allows the initial radial clearance to be fixed during pump assembly by placing shim material between the pump body and the clamping flange 50 of each cylindrical housing 44.

Located between the bearings 48 and the end wall 52 of the cylindrical housing 44 are a spacer ring 54 and a spring 56. The bearings 48 are retained in the housing 44 by means of a clamping ring 58 such that a preload on the spring 56 is set for the running load condition (and input power of the pump). The end wall 52 extends radially inwardly towards a collar 60 which forms part of an assembly for fastening the cylindrical housing 44 to the shaft.

In use, compression work done by the pump 10 results in an axial load tending to move the rotors 34, 36 in a direction from the outlet 22 towards the inlet 20. The axial load acts against the springs 56 to cause the rotors 34, 36 to move in an axial direction, thereby changing the radial clearance between the threads 38, 40 of the rotors 34, 36 and the stator in proportion to the axial load. By changing the characteristic spring rate, the input power of the pump can be tailored to a specific

application over the speed range of the pump. Where there is any difference in axial load on the rotors 34, 36, the connector 46 rigidly connecting the two cylindrical housings 44 together ensures that both rotors are repositioned simultaneously, avoiding any interference which may occur between the threads 38, 40 of the rotors 34, 36 should they become misaligned with respect to each other.

In a variation of the embodiment of Figures 1 to 7, the axial positions of the rotors are controlled by pneumatically controlled pistons. Movement of the pistons may be controlled by a control mechanism which may include a force sensor which detects the axial load on a given rotor axis and causes a reactive force to be applied by means of the pistons. In addition, the controller may be configured to allow independent movement of the pistons and hence the rotors for other purposes, for example rotor cleaning.

Figure 8 illustrates a second embodiment of a screw pump having active control of the positions of the rotors within the stator. Similar to the first embodiment, the pump 70 includes a pump body 72 defining a pumping chamber 74, fluid inlet 76 and fluid outlet 78. The pump 70 further includes a first shaft 80 and, spaced therefrom and parallel thereto, a second shaft 82. First bearing assemblies 84 are provided for supporting the upper ends (as shown in Figure 8) of the shafts 80, 82, and second bearing assemblies 86 located within bearing housing 88 are provided for supporting the lower ends of the shafts 80, 82. The shafts 80, 82 are adapted for rotation within gearbox 83 about the longitudinal axes in a contra-rotational direction. One of the shafts 80 is connected to a drive motor (not shown) via a drive mechanism, the shafts being coupled together by means of timing gears 90 so that in use the shafts 80, 82 rotate at the same speed but in opposite directions.

A first rotor 92 is mounted on the first shaft 80 for rotary movement within the chamber 74, and a second rotor 94 is similarly mounted on the second shaft 82. Each of the two rotors 92, 94 has a first part proximate the inlet 76 having a

generally cylindrical shape and a second part proximate the outlet 78 having a tapered shape. Each rotor has a helical vane or thread 96, 98 respectively formed on the outer surface thereof, the threads intermeshing as illustrated.

The screw threads 96, 98 of the rotors 92, 94 have, on the first part of each rotor, a substantially constant outer diameter and, on the second part of each rotor, an outside diameter which tapers decreasingly towards the outlet 78 of the pump 70. The inner surface of the pumping chamber 74, which acts as a stator during use of the pump, is conformingly shaped to the shape of the outer diameters of the rotors.

In order to control the axial position of the rotors 92, 94 within the pumping chamber 74, the pump 70 includes a servo motor 100 which rotates a lead screw 102 attached thereto. The lead screw 102 engages a conformingly-threaded aperture 104 in the bearing housing 88 so that the bearing housing 88 acts a piston, moving axially relative to the pumping chamber 74 to control the rotor to stator clearance over the tapered section of the pump 70. Actuation of the servo motor 100 can be controlled by any suitable mechanism. For example, one or more sensors (not shown) can provide to the motor 100, or to a controller thereof, signals indicative of back pressure, exhaust temperature, power consumption and/or inlet pressure for use in controlling the axial position of the rotors during use of the pump 70.

The mechanism for axially moving the rotors relative to the stator in this second embodiment can, of course, be used to move wholly tapered rotors as used in the first embodiment.

In Figure 8, the mechanism for axially moving the rotors relative to the stator is located at the low pressure (inlet) end of the pump. However, this mechanism could alternatively be located at the high pressure (exhaust) end of the pump.

CLAIMS

1. A screw pump comprising a stator housing first and second externally threaded rotors adapted for counter-rotation within the stator, and means for actively controlling the axial position of the rotors within the stator during use of the pump.

2. A screw pump according to Claim 1, wherein the control means comprises means for effecting or resisting axial movement of the rotors in response to an axial load generated in the rotors during operation of the pump.

3. A screw pump as claimed in Claim 1 or Claim 2, wherein each rotor is mounted on, or integral with, a respective shaft rotatably mounted within the pump.

4. A screw pump according to Claim 3, comprising a bearing assembly for rotatably supporting the shafts relative to the stator, the control means comprising means for moving the bearing assembly relative to the stator.

5. A screw pump according to Claim 4, comprising a piston engaging the bearing assembly, the moving means being arranged to move the piston relative to the stator to control the axial position of the rotors.

6. A screw pump according to Claim 5, wherein the moving means comprises a motor adapted to rotate a drive shaft which engages the piston so as to axial move the piston relative to the stator with rotation of the drive shaft.

7. A screw pump according to Claim 6, wherein the drive shaft comprises a lead screw which passes through a conformingly-threaded aperture in the piston.

8. A screw pump according to any of Claims 5 to 7, wherein the piston comprises part of a housing for the bearing assembly.

9. A screw pump according to any of Claims 1 to 3, wherein the control means comprises a piston actuated by a control device so as to control the axial position of the rotor.

10. A screw pump according to Claim 9, wherein the control device is responsive to an input power and resulting axial load on a rotor to cause the piston to actuate so as to control the axial position of the rotor.

11. A screw pump according to Claim 4, wherein the bearing assembly is free to move in an axial direction within a housing, the control means comprising a spring mechanism arranged with respect to a rotor such that when the rotor is subjected to an axial load, the spring mechanism compresses or extends causing an axial reactive load, whereby to maintain a constant axial position of the rotor over time.

12. A screw pump according to Claim 11 wherein the spring mechanism comprises a setting spring positioned in the housing between the bearing assembly and an end surface of the housing.

13. A screw pump according to Claim 11 or Claim 12, wherein the housing is a cylindrical housing having an end surface extending radially inwardly toward the rotor.

14. A screw pump according to any of Claims 11 to 13, wherein the spring mechanism is selected such that the maximum axial load to which a rotor is likely to be subjected does not exceed the elastic limit of the spring mechanism.

15. A screw pump according to any preceding claim, wherein the control means is operable to allow non-synchronous as well as synchronous axial displacement of the rotors.

16. A screw pump according to any preceding claim, wherein at least part of the screw threads of the rotors have an outside diameter which tapers decreasingly in a direction from the pump inlet to the exhaust of the pump.

17. A screw pump according to Claim 16, wherein only part of the screw threads of the rotors has an outside diameter which tapers towards the exhaust of the pump.

ABSTRACT

A screw pump comprises a pair of rotors 2a, 2b each carrying an external screw thread, the pair of rotors being rotatably mounted in a stator 1 and arranged such that, in operation, the screw threads of the rotors intermesh as the rotors rotate in opposing directions. Means are provided for actively controlling the axial position of the rotors within the stator during use of the pump.

(Figure 1)

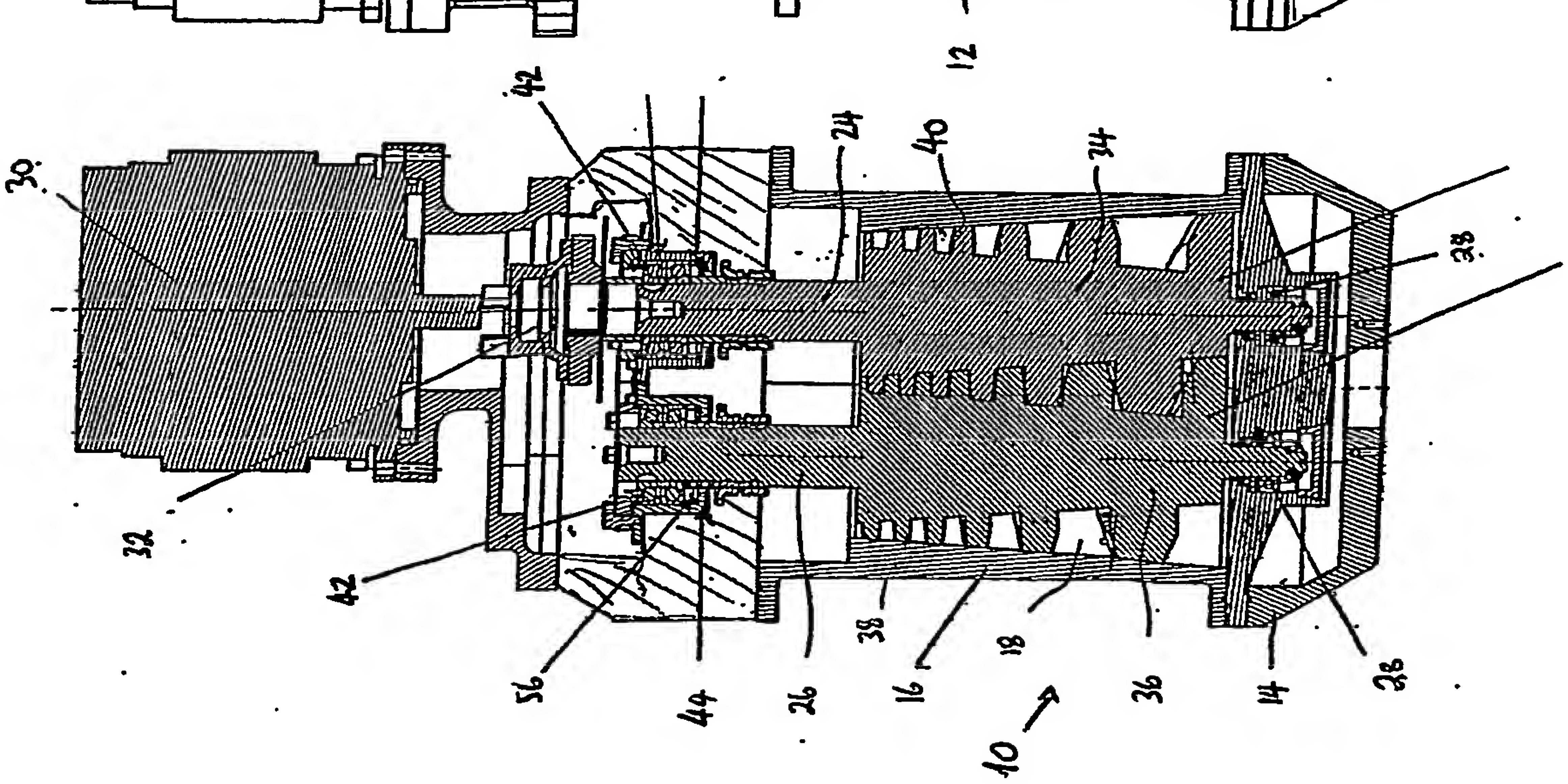

FIG. 1

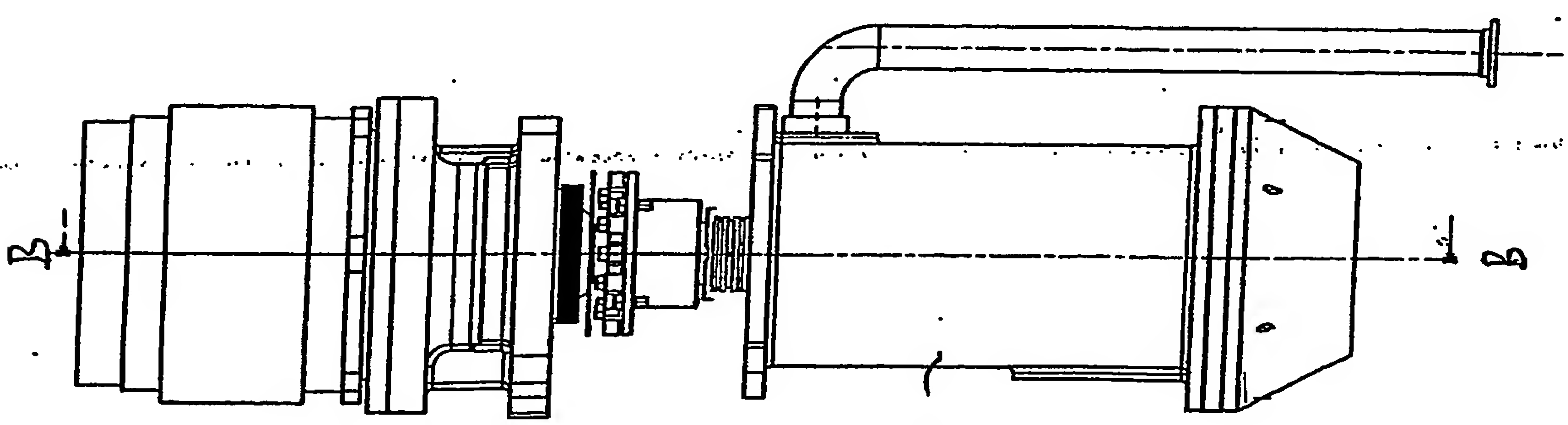

FIG. 2

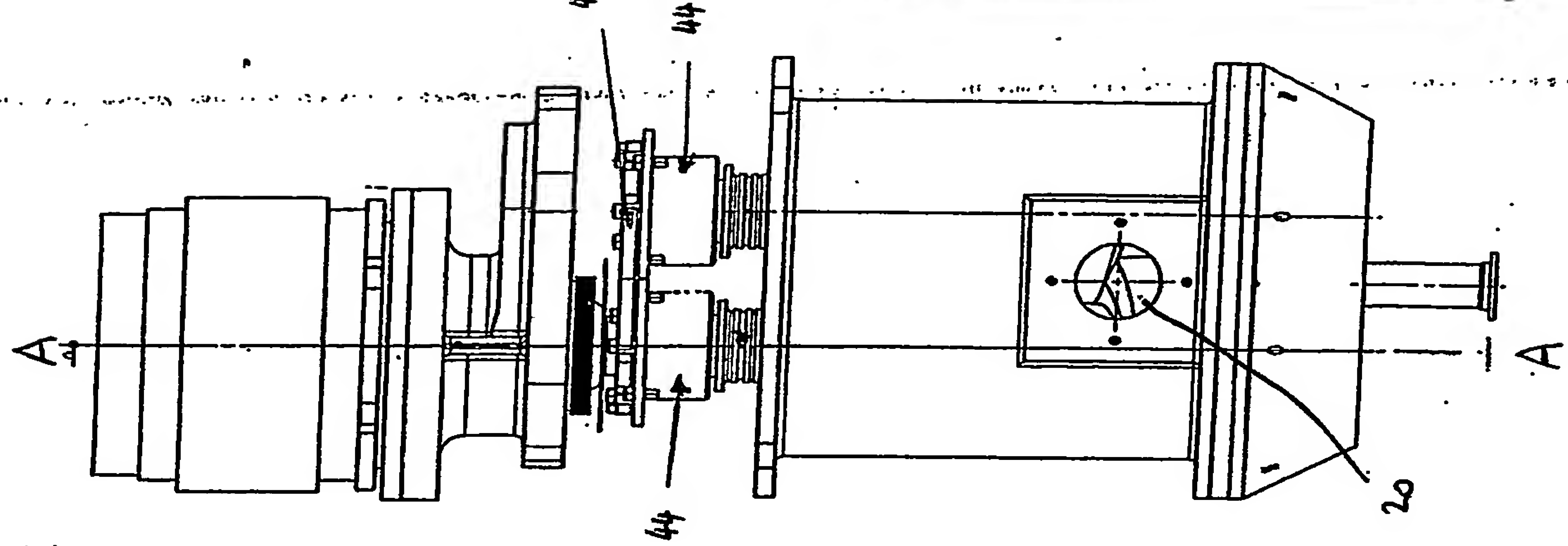

FIG. 3

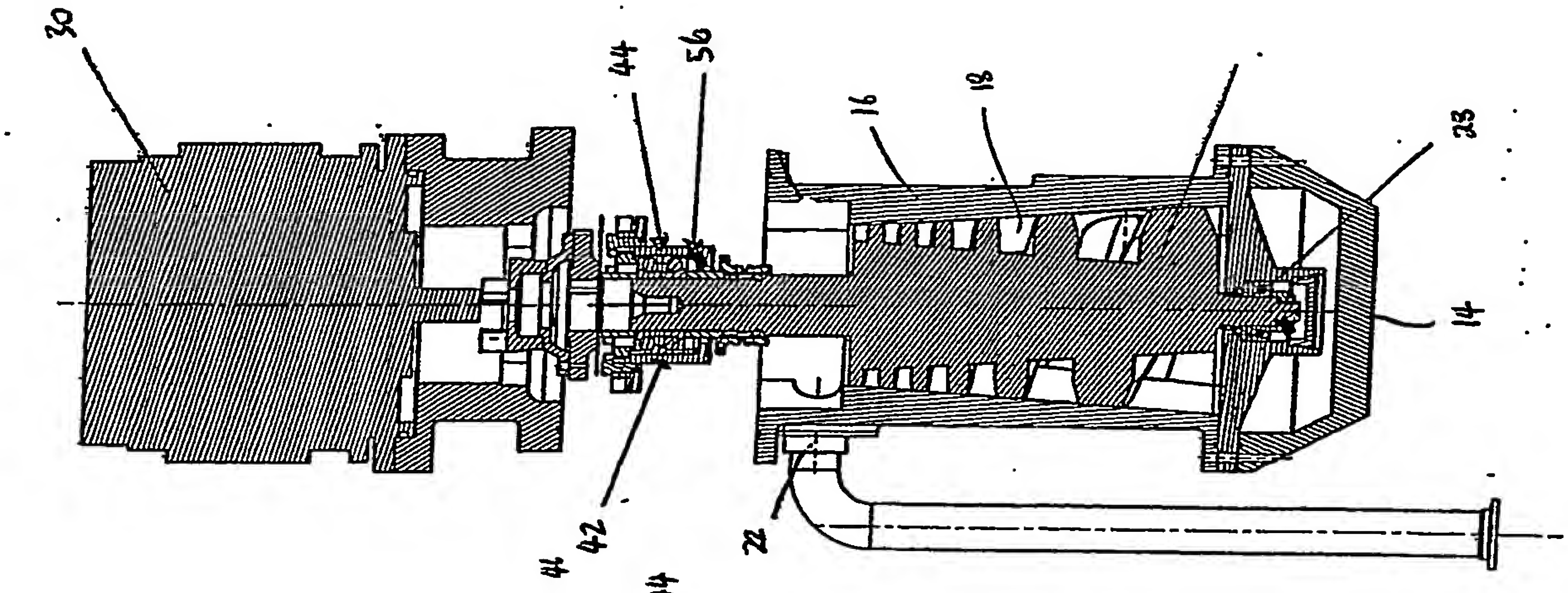

FIG 4

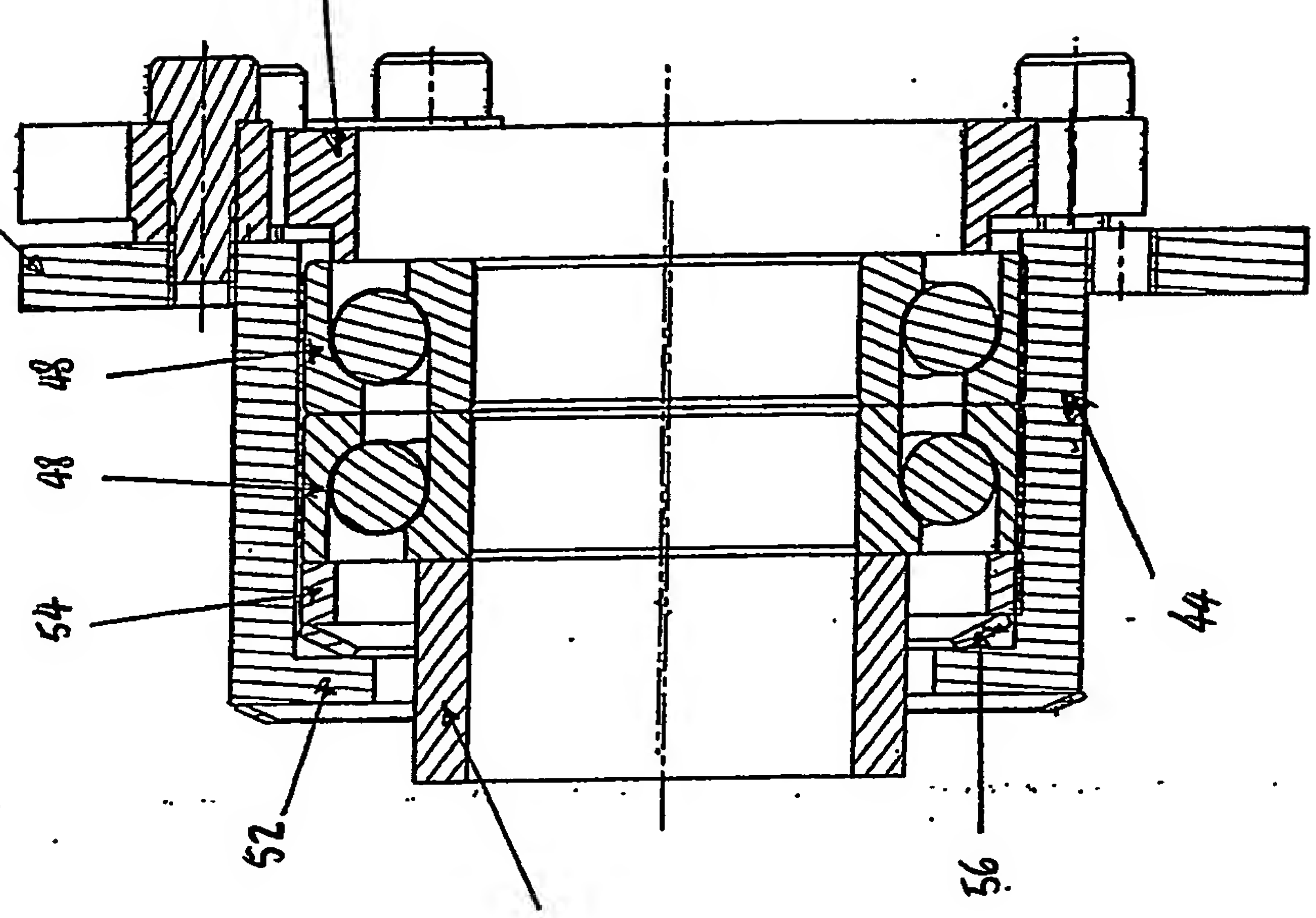

Fig.6

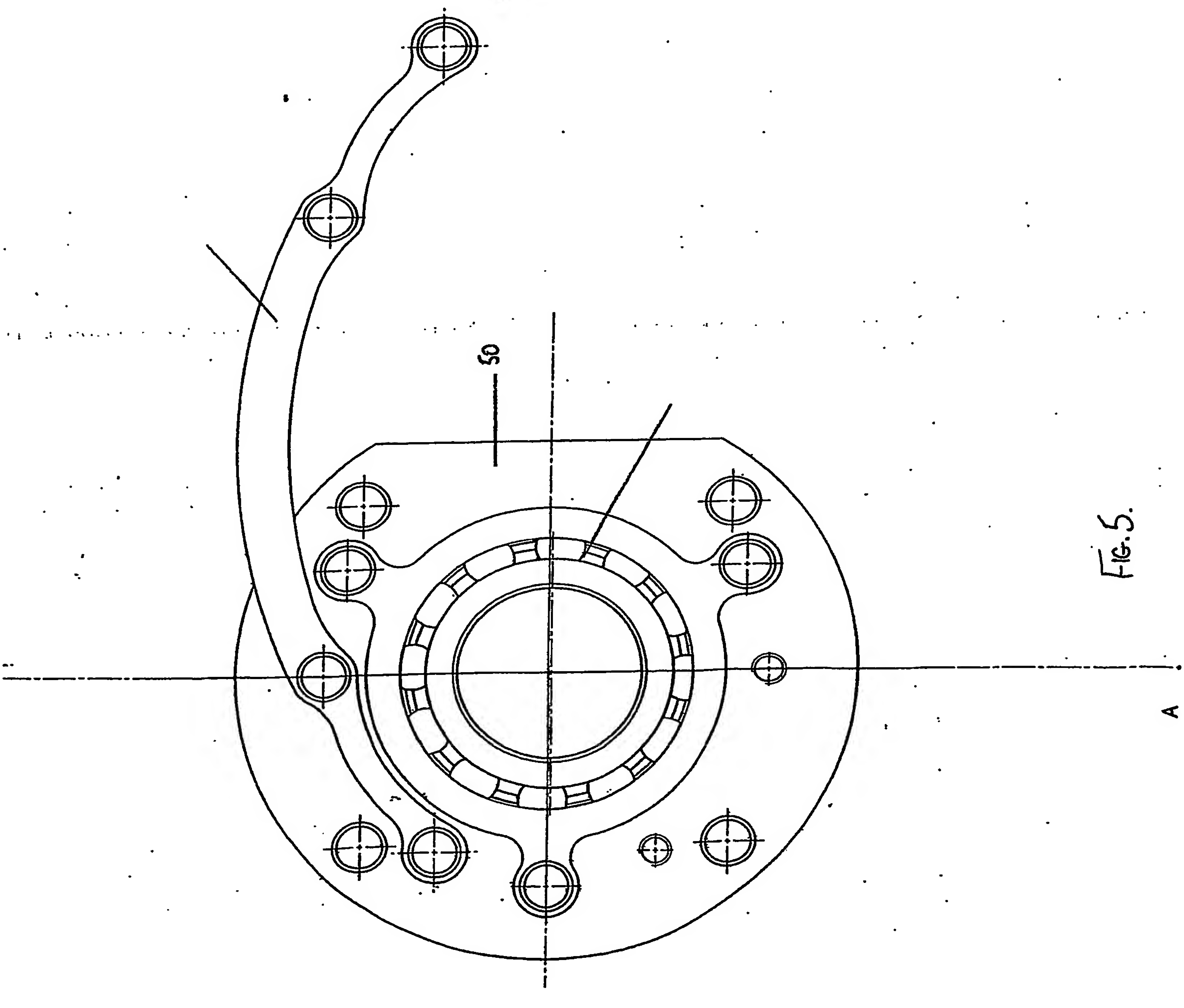

Fig.5.

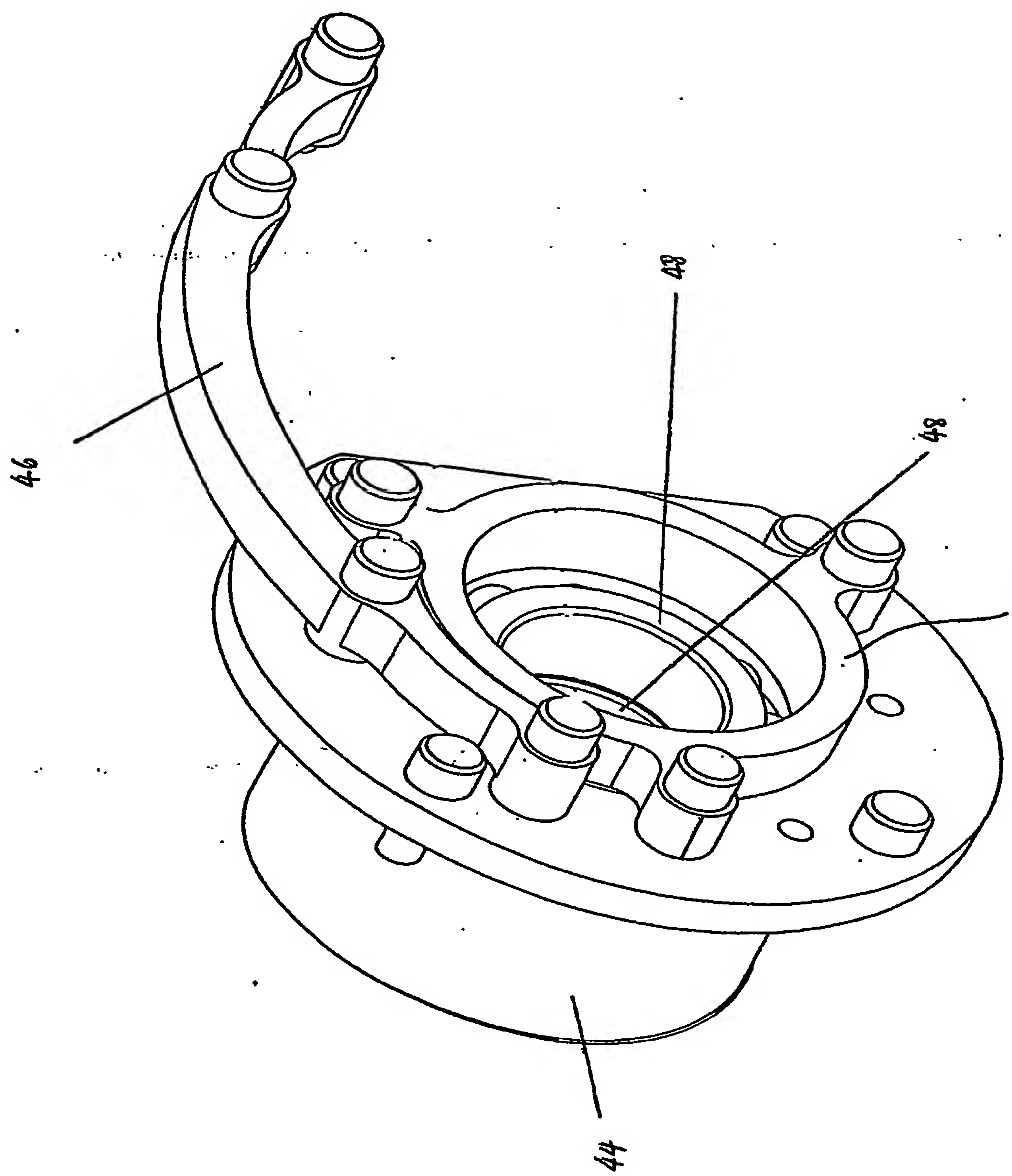

FIG. 7

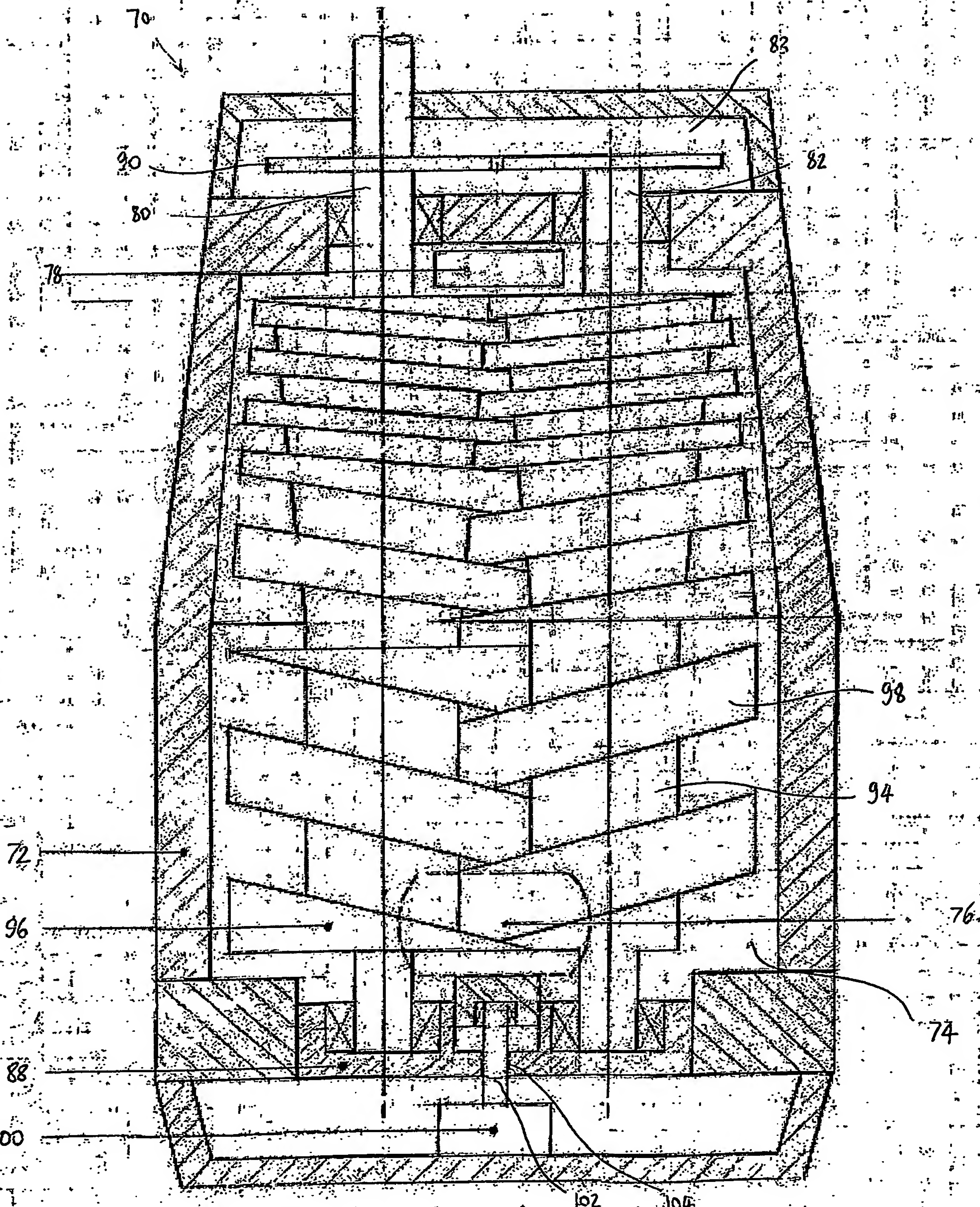

FIGURE 8